

EXPERIMENTAL INVESTIGATION OF CONICAL COMPRESSOR WITH VAPOUR COMPRESSION REFRIGERATION SYSTEM USING HYDROCARBON AS A REFRIGERANT

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ABSTRACT

This paper presents experimental analysis of newly designed conical rotary compressor prototype with refrigerant R134a and two hydrocarbon mixtures of propane and isobutane in varying proportion. The experimental tests are carried out for maximum pressure ratio and maximum cooling capacity. Compressor tests are also conducted for varying pressure ratios. For each test, the refrigerant mass flow rate, electrical power consumption, and the suction and discharge temperature and pressure are recorded. Using the experimental data, the compressor volumetric and isentropic efficiencies and the corresponding COP are calculated. The volumetric efficiency ranges from 44% to 78% and the isentropic efficiency varies from 59% to 80%. The maximum pressure ratio achieved is 4.9. The cooling capacity for maximum pressure ratio is observed as 240 w for R134a, 630 w for mixture of R290 and R600a in the ratio of 50:50 and 632 w for R436a (composed of R290 and R600a in the ratio of 56:44 by weight). Because of its small size and less weight, it can potentially be used in a vapor compression refrigeration system for cooling applications.

KEYWORDS: Comparison, Conical Compressor, Performance, Refrigerants, Rotary

INTRODUCTION

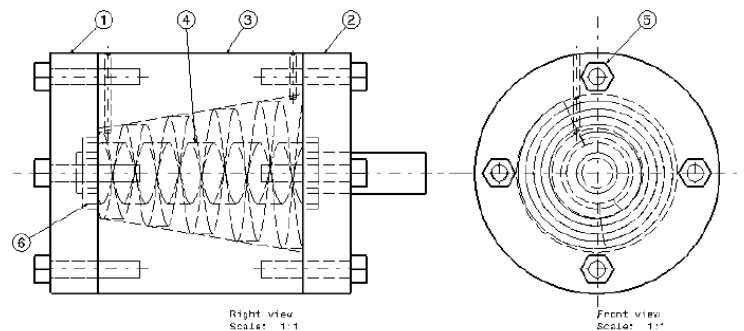
The conversion in the sphere of refrigeration from CFC and CFC related technologies to hydrocarbon technologies is a remarkable piece of industrial history. Hydrocarbons are excellent refrigerants, both for their energetic efficiency and their environmental compatibility witnessed by zero ODP and negligible GWP [1]. Unfortunately, they are highly flammable and for this reason they are not widely applied. A large number of development tasks have been completed. An extensive range of compressors and equipment is already available for the various alternative refrigerants. Most important criteria in this concern are flammability, thermodynamic properties and global warming potential [2].

A fair comparison of the performance merits of different refrigerants for a given application requires systems in which all components are optimized for the given refrigerant. Selecting a suitable expansion device presents no difficulty, while some care is needed to specify appropriate connection tubes. A more challenging task is to provide heat exchangers and compressors that are optimized individually for each refrigerant. In particular, providing optimized compressors may prove to be the most difficult task because the compressors should be sized properly to assure equal cooling capacity for each system. In order to achieve the aim of use of compressor for different refrigerants, the basic modification is necessary.

In various types of existing compressors type of compressors much progress has been made in the past few decades and it is foreseen that only incremental improvements will be possible in the future. Therefore, in order to create a potential leap in performance, a new compressor design, named 'Conical Rotary compressor' has recently been introduced, in which the conical shape type rotor concept effectively reduce frictional losses. This paper presents the performance of Conical Compressor, where it has undergone preliminary experimental investigations using R134a and hydrocarbon mixture as a refrigerant. The important external dimensions of the compressor and its specifications are given in Table 1. The detail sketch of this compressor is as shown in figure 1

Table 1: External Dimensions of Conical Compressor

Sr.No.	Geometrical Parameters	Dimensions
1	Diameter(mm)	118.6
2	Length (mm)	99
3	Displacement(CC)	30
4	Weight (Kg)	7
5	Speed(RPM)	1000



PART NO.	PART NAME	MATERIAL	QUANTITY
6	BEARING	STD.	2
5	NUT & BOLT	M.S.	8
4	ROTOR	ALUMINIUM	1
3	STATOR	M.S.	1
2	SUCTION END PLATE	ALUMINIUM	1
1	DISCHARGE END PLATE	ALUMINIUM	1

Figure 1: Detail Sketch of Conical Compressor

THEORETICAL ANALYSIS

The system has been developed for theoretical computational analysis of the cooling system in order to obtain estimates of the process of system operation as well as its performance. For this analysis the dedicated software, namely REFPROP 9.0 is used for evaluation of thermodynamic and thermo physical properties of refrigerants[3]. Low GWP Hydrocarbon mixtures in proportion of 50:50 and 56: 44 of propane(R290) and isobutane(R600a) respectively were selected for experimentation. Their performance with conical compressor in vapour compression refrigeration system has investigated. The physical properties of selected refrigerant are as shown in table 2. In the refrigeration system, the representative performance characteristics are compressor power (W_c), refrigerating effect (Q_e) and Coefficient of

Performance (COP).

Table 2: Physical Properties of Refrigerants

Sr. No.	Refrigerant	Boiling Point @ 1 atm (K)	Freezing Point (K)	Critical Temp (K)	Critical pressure (bar)	ODP	GWP
1	R134a Tetrafluoroethane (HFC)	247.00	176.55	374.25	40.67	0	1160
2	R290 Propane (HC)	231.07	85.49	369.83	42.1	0	3
3	R600a Iso butane (HC)	261.48	113.55	138.15	36.45	0	20

Figure 2 shows a real thermodynamic model of a vapour compression refrigeration system, which consist essentially a compressor, an evaporator, a condenser and a capillary tube. These components are connected by pipelines in which a refrigerant with suitable thermodynamic properties circulates. The theoretical analysis for the performance parameters are calculated as follows [4].

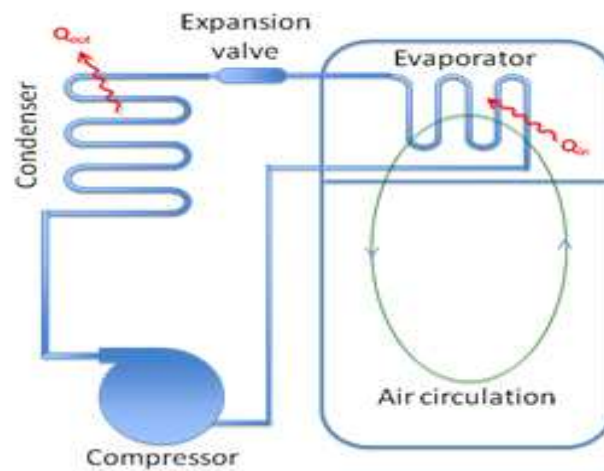


Figure 2: Real Thermodynamic Model of a Vapour Compression Refrigeration System

- i) Compressor pressure ratio (Pr) is given as:

$$P_r = P_{dis} / P_{suc}$$

Where, P_{dis} = Refrigerant vapour pressure at the compressor discharge

P_{suc} = Refrigerant vapour pressure at the compressor suction.

From P-h chart as shown in figure 4 varying discharge pressure

$$P_{dis} = P_3 \text{ (Condenser pressure)}$$

- ii) Volumetric efficiency = $m V_{suc} / \text{RPM} * \text{Vol.}$

The suction specific volume (V_{suc}) is calculated as a function of suction temperature and pressure. The Compressor displacement volume (Vol.) is obtained from the manufacturer's specifications given in Table 1.

- iii) Isentropic efficiency = $m(h_{dis} - h_{suc}) / W_{actual}$

The suction enthalpy (h_{suc}) is calculated as a function of the suction temperature and pressure. The isentropic discharge enthalpy (h_{dis}) is calculated by assuming constant entropy for the compression process.

iv) Refrigerating Effect (Q_{theo}) = $m (h_{\text{suc}} - h_{\text{evaporator,in}})$

The theoretical cooling capacity is calculated by assuming that the compressor is a part of a hypothetical vapor compression refrigeration system. The following assumptions are made:

- The pressure drops in the condenser and the evaporator of the hypothetical system are neglected.
- The sub cooling and superheating are neglected.
- An isenthalpic pressure drop is assumed in the throttle device.

The evaporator inlet enthalpy is calculated as a function of the condenser outlet enthalpy and the suction pressure. The condenser discharge enthalpy is calculated as a function of the compressor discharge pressure [5].

v) Coefficient of performance (COP) is calculated as

$$COP_{\text{actual}} = Q_{\text{actual}} / W_{\text{actual}}$$

EXPERIMENTAL SETUP

Compressor load stand was designed for testing of the compressors. A detailed schematic diagram of the vapour compression refrigeration system with hot gas bypass arrangement is shown in figure 3. The main idea behind this concept is to use an intermediate pressure for the test stand by condensing a fraction of the refrigerant flow. Using this stable pressure, discharge pressures are controlled by using appropriate metering valves in the discharge line and bypass line. The p-h diagram shown in Figure 4 is frequently used in the analysis of vapour compression refrigeration cycle. The system was instrumented with two pressure gauges with accuracy of $\pm 1\%$ at the inlet and outlet of the compressor manifold for measuring the suction and discharge pressures. The temperature of the refrigerant at four different points was measured with copper-constantan thermocouples with accuracy of $\pm 1^\circ\text{C}$. The energy consumption of the refrigeration system was measured using energy meter with accuracy of $\pm 3\%$. The mass flow rate of the refrigerant was measured using a flow meter with $\pm 0.1\%$ accuracy installed in the liquid line between the condenser and the capillary tube. Charging ports were installed at the inlet of the compressor for charging and recovering the refrigerant. The evacuation of moisture in the system was also carried out through the port. The refrigerant R134a and mixture of hydrocarbon was used in Vapour compression refrigeration system. The thermodynamic properties of refrigerant used have been considered for theoretical and experimental analysis. [6]

Experimental tests on the conical compressor were conducted for its maximum capacity and maximum pressure ratio. The compressor performance was also analyzed with vapour compression system for different pressure ratios by keeping constant suction pressures. The parameters are directly measured using the instrumentation as suction pressure and temperature, discharge pressure and temperature, refrigerant mass flow rate(m), electrical power consumed by the compressor(W_{actual}) and refrigeration effect i.e. heater input given to the calorimeter.(Q_{actual}).The system was charged with the help of charging system and evacuated using vacuum pump to remove the moisture. After charging the refrigerant, data were collected at different discharge pressure.

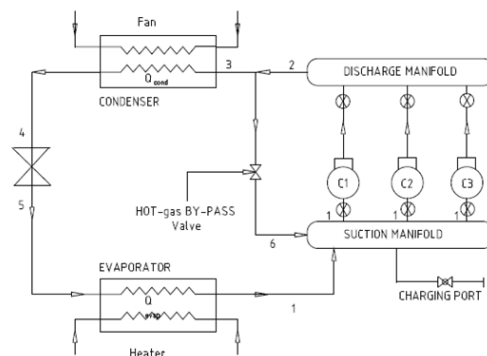


Figure 3: Schematic Diagram of Compressor Load Stand

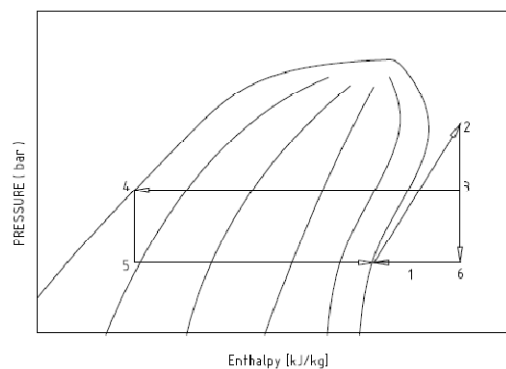


Figure 4: Vapour Compression Refrigeration Cycle on P-H Diagram

RESULTS

The conical compressor performance parameters defined in the previous section are plotted as a function of the compressor pressure ratio for the different discharge pressures. Figure 5 shows the variation of volumetric efficiency with pressure ratio for different refrigerants. The volumetric efficiency decreases slightly with increase in the pressure ratio and ranges from 44% to 78% for maximum pressure ratio whereas 6 % to 7% more for hydrocarbon mixture. The performance measurements indicate that the volumetric efficiency is less in case of refrigerant R134a whereas slight difference is observed in case of hydrocarbon mixture (50:50) and R436a (56:44). Figure 6 presents the variation of the isentropic efficiency of the compressor as a function of the pressure ratio. Although a clear trend is not observed, it can be seen that the compressor working influence on the isentropic efficiency. The overall isentropic efficiency varies in the range of 59% to 80%. The isentropic efficiency is less for higher compression ratio.

The refrigeration capacity of the system is plotted as a function of the pressure ratio in Figure 7. The refrigeration capacity decreases with an increase in pressure ratio (the evaporator inlet enthalpy decreases with increase in the compressor discharge pressure, which reduces the enthalpy difference available in the evaporator resulting in a drop in the observed drop in cooling capacity). In addition, the refrigerant used in a system has a significant influence on the cooling capacity. The conical compressor with R436a has more cooling capacity than other refrigerants used for experimentation. The highest cooling capacity of 815w is observed in case of refrigerant R436a. The cooling capacity for conical compressor has been ranges from 240 w to 525 w for R134a whereas 630 w to 815 w for hydrocarbon mixtures. Figure 8

shows the power consumption of conical compressor for different refrigerants. As discharge temperature is more for hydrocarbon mixture, comparatively power consumption is also more compared with R134A.

Finally, the system coefficient of performance (COP) is shown in Figure 9 as a function of pressure ratio. The COP decreases rapidly as the pressure ratio increases. The performance measurements indicate that the COP is more for higher compression ratio in case of hydrocarbon mixture than R134a. For the given set of tests, the COP varied from 1.08 to 4.67.

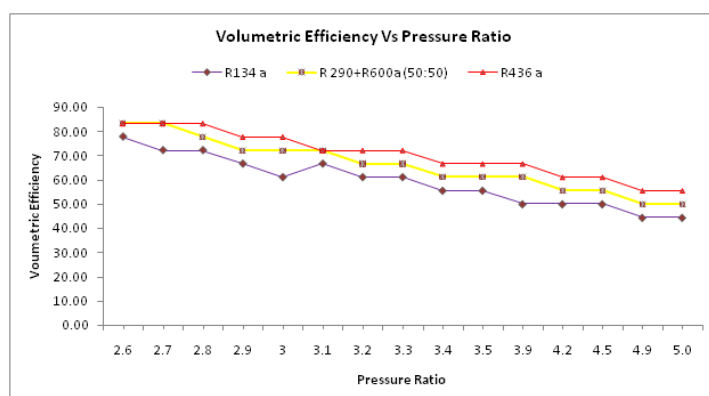


Figure 5: Variation of Volumetric Efficiency with Pressure Ratio

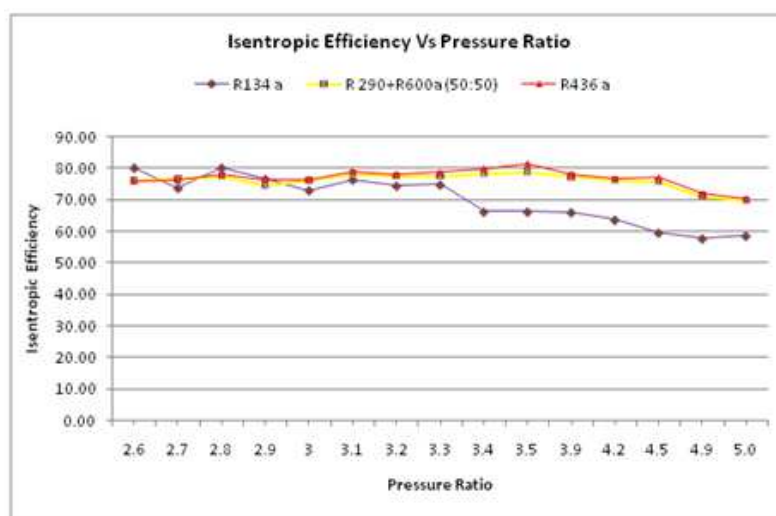


Figure 6: Variation of Isentropic Efficiency with Pressure Ratio

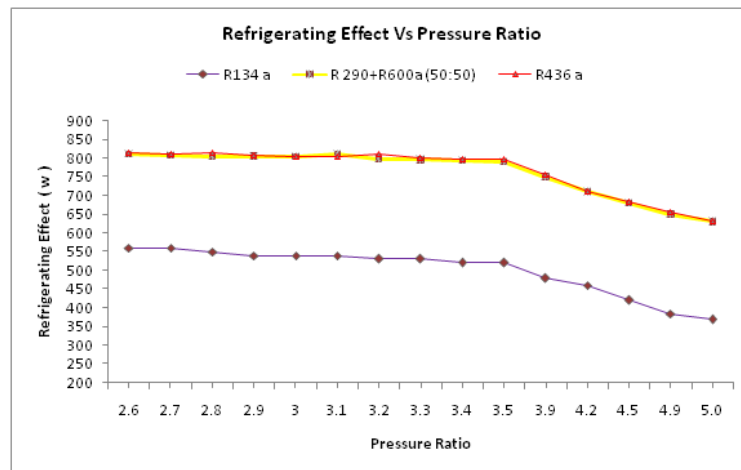


Figure 7: Variation of Refrigerating Effect with Pressure Ratio

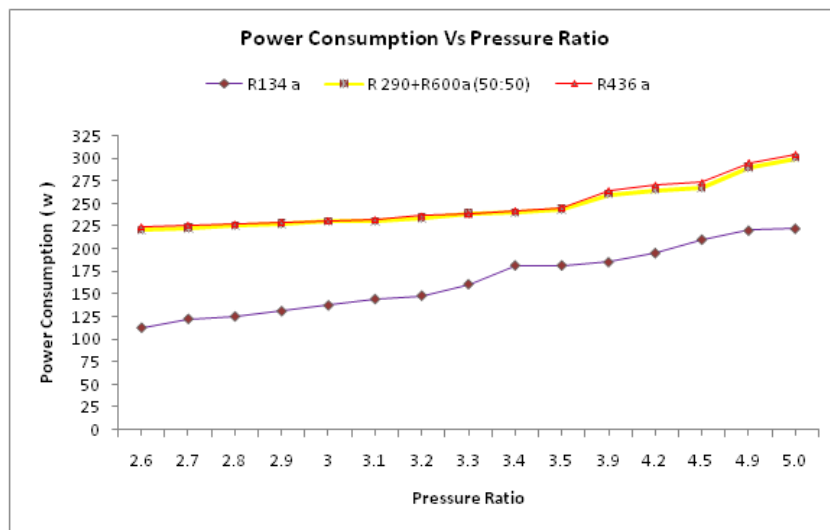


Figure 8: Variation of Power Consumption with Pressure Ratio

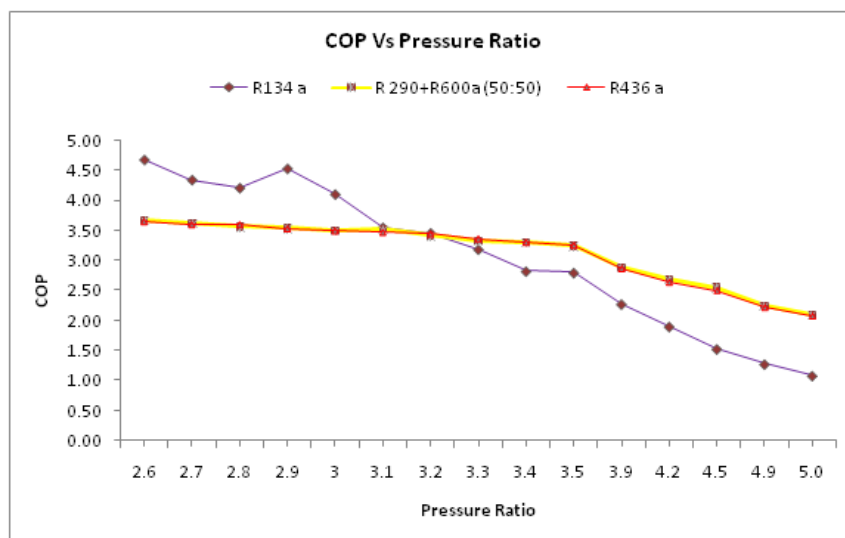


Figure 9: COP Variations with Pressure Ratio

CONCLUSIONS

A Conical rotary compressor prototype is experimentally tested using different refrigerants. The tests were conducted for maximum capacity and maximum pressure ratio achieved in newly designed conical compressor. The experimentation was also done for different pressure ratios with varying discharge pressures. Important performance parameters were either directly measured or calculated using basic thermodynamic relations. The experimental results indicate that the compressor renders good volumetric and isentropic efficiencies. The conical compressor performance has been compared to that of two hydrocarbon mixtures which is substitute for R134a for refrigeration application. The comparison indicates that the compressor investigated here performs better in terms of the efficiencies and cycle COP. With its small size and low weight, the compressor has the potential for use in vapor compressor refrigeration systems for cooling application such as water cooler, freezer and bottle cooler. The study shows that compressor has also used for different refrigerants without any modification.

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